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Step-by-step design methodology for efficient Stirling-type pulse tube refrigerator

Taekyung Ki*, Sangkwon Jeong¹

Cryogenic Engineering Laboratory, Division of Mechanical Engineering, School of mechanical, Aerospace and Systems Engineering, Korea Advanced Institute of Science and Technology, 373-1, Guseong-dong, Yuseong-gu, Daejeon 305-701, South Korea

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ABSTRACT

This paper presents an improved design method and performance prediction of Stirling-type pulse tube refrigerator (PTR). The paper is composed of three parts and each content is as follows: 1) specific design and fabrication of each component, 2) modeling and configuration optimization of Stirling-type PTR, and 3) experimental verification. Each part provides a step-by-step procedure to easily understand the design process for developers and researchers who want to develop efficient Stirling-type PTRs. Through this proposed design process, an efficient Stirling-type PTR is carefully designed for cooling capacity of 8 W at 50 K. The fabricated Stirling-type PTR possesses the cooling capacity of 7 W at 50 K and the Carnot efficiency of 11.7% at 56.3 K. The experimental results show the proposed design methodology fulfills the required performance of Stirling-type PTR with desirable accuracy. The design process presented in this paper can help developers and researchers to design an efficient Stirling-type PTR.

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Méthodologie fondée sur une approche progressive pour la conception de refroidisseurs performants de type Stirling à tube à pulsation

Mots clés : Cryorefroidisseur ; Conception ; Cycle ; Efficacité ; Tube à pulsation ; Stirling

1. Introduction

Cryocoolers are widely used in many application areas such as space exploration, military, medicine, energy industry and laboratory. A pulse tube refrigerator (PTR) has several inherent advantages compared to Stirling and Gifford-McMahon (GM)

cryocooler such as no moving parts in the cold stage, low manufacturing cost, high frequency of operation and long operation life (Richardson and Evans, 1997). There have been great research interest and development efforts so far due to such advantages. The previous researches are mostly concerned about the development and performance analysis of

* Corresponding author. Tel.: +82 42 350 3079; fax: +82 42 350 8207.

E-mail addresses: kingcandy@kaist.ac.kr (T. Ki), skjeong@kaist.ac.kr (S. Jeong).

¹ Tel.: +82 42 350 3039; fax: +82 42 350 8207.

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Nomenclature			
A	Area [m ²]	amb	Ambient
D	Diameter [m]	C	Cold-end
e	Specific gas energy [J kg ⁻¹]	H	Hot-end
f	Friction factor [–]	h	Hydraulic
L	Length [m]	m	Mesh
M	Mass [kg]	Q	Heat [W]
P	Pressure [Pa]	Q _{sw}	Heat flow in surface [W m ⁻²]
Greek symbol		q	Heat flux [W m ⁻¹]
λ	Pitch of mesh [m]	T	Temperature [K]
ρ	Density [kg m ⁻³]	t	time [s]
η	Carnot Efficiency [–]	u	Velocity of gas [ms ⁻¹]
Subscript		W	Input power [W]
		r	Regenerator
		w	Wire

each component in conjunction with the simulation method for improving its cooling power and efficiency.

Researches on a compressor are focused on the attainment of high efficiency and high frequency in a smaller size. A linear compressor is one of the compressors for satisfying those demands. The PTR driven by a linear compressor is usually named as a Stirling-type PTR. The performance of Stirling-type PTR can be improved by understanding on the coupled hydraulic behavior between the cold part and linear compressor. Since losses generated in a regenerator and pulse tube directly affect the cooling capacity and efficiency of the PTR, the related researches have been conducted to reduce the loss and analyze the cause. From the results of preceding researches, the shape of the regenerator and pulse tube is optimally designed. Taylor et al. (2008) suggested a computational fluid dynamics (CFD) analysis to optimally design the configuration of pulse tube. For satisfying the required heat load of each heat exchanger in the PTR, heat exchangers of various types and configurations are developed and applied. Lewis et al. (2009) used a quadrant mesh-type heat exchanger to reject heat in the heat exchanger having a large diameter. Ki and Jeong (2010a) developed the design method of slit-type heat exchanger and used it for the on-board cryocooler. The approaches for theoretical study and analysis on PTR are mostly conducted by numerical simulation. Most of the available references are helpful to understand physical mechanism and design principle of PTR.

However, since there is no systematic design method explained by a step-by-step procedure, most of developers and researchers who need straightforward design process for developing the PTR satisfying the required cooling capacity and efficiency cannot fulfill their demands easily. As a result, lots of extra efforts are made for searching and arranging the references.

In this paper, the systematic design process for developing efficient Stirling-type PTR is suggested. The process is composed of three parts, and the content is as follows: 1) specific design and fabrication of each component, 2) modeling and configuration optimization of Stirling-type PTR, and 3) experimental verification. Each part provides a step-by-step procedure for manufacturing an efficient Stirling-type PTR. This proposed process can reduce unnecessary spending of time and money in development of Stirling-type

PTRs. It can contribute to increasing its usefulness and the practicality of Stirling-type PTR in various fields.

2. Specific design and fabrication of each component

2.1. Performance requirement of PTR

Since the configuration of cryocooler is sometimes determined by the performance and application purpose, the first step of design process is to determine required performance of the cryocooler in the applied field. We determine the cooling capacity as 8 W at 50 K, and the configuration of the Stirling-type PTR is considered as the in-line type pulse tube for the potential application in HTS (High Temperature Superconductors) system. The schematic diagram is shown in Fig. 1.

2.2. Linear compressor

The required electric power of linear compressor must be estimated for satisfying the demanded cooling capacity. Eq. (1) (Kittel, 2008) shows the relation between the cooling capacity and the compressor input power.

$$\frac{Q_C}{W} = \eta \frac{T_C}{(T_H - T_C)} \quad (1)$$

When the Carnot efficiency is set by 10% at 50 K, the calculated compressor power is 400 W. The linear motor (1S175M Star motor, CFIC Inc.) is used to supply enough power more than 400 W in this paper.

A housing and cylinder are designed to be able to sustain pressure of 4 MPa with finite element analysis. The housing material is stainless steel 304, and a bounce volume of the linear compressor is carefully designed because it greatly affects the resonance frequency of the PTR. The cylinder material is selected as aluminium 7075 for effective cooling during compression process. A clearance gap between the piston and cylinder is carefully adjusted by CFIC Inc., and the measured gap size is 0.028 mm. Since the single-piston type linear compressor is used, the vibration of linear compressor must be alleviated by a dynamic absorber. The dynamic absorber is designed and

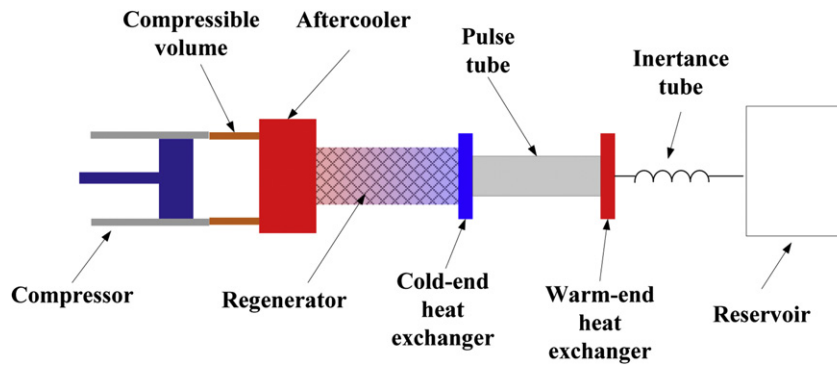


Fig. 1 – Schematic diagram of Stirling-type PTR.

fabricated by referring our previous research (Ki and Jeong, 2010a). Fig. 2 shows the fabricated linear compressor.

2.3. Heat exchanger

The heat exchanger considerably affects the performance of cryocoolers, so the heat exchanger must be carefully designed. In the Stirling-type PTR, three heat exchangers (aftercooler, cold-end heat exchanger (CHX) and warm-end heat exchanger (WHX)) are employed. The aftercooler is used for removing the heat generation from compressor, the CHX for cooling the heat load, and the WHX for removing the heat generation between pulse tube and inertance tube.

In this paper, slit-type heat exchangers are used as the heat exchangers. The slit-type heat exchanger has several advantages compared to mesh-type heat exchanger. It has no thermal contact resistance problem that is usually encountered in a mesh-type one because the flow path of slits is directly made in copper block by electrical discharge machining (EDM). In addition, if the cryocooler has different flow cross-sections around heat exchangers, the slit-type heat

exchanger can make smooth flow transition. The appropriate heat path of radial direction is especially favorable in a large scale and high power cryocooler. For applying the slit-type heat exchanger in the design process of the Stirling-type PTR, the design method (Ki and Jeong, 2010a) is used. From this process, the configuration of the slit and temperature of inlet and outlet gas in the slit-type heat exchangers are determined, and these parameters are used as the configuration and boundary condition of the heat exchanger in the simulation model of the Stirling-type PTR. The structural configurations of designed slit-type heat exchangers are listed in Table 1. Finally, the pressure drop characteristic of the slit-type heat exchanger investigated from the previous research (Ki and Jeong, 2010b) is used as the friction factor and the local (entrance and exit) loss coefficients of the heat exchanger in the simulation model of the Stirling-type PTR. From these processes, thermal and hydraulic characteristics of heat exchangers can be precisely obtained and fully considered in the simulation model.

2.4. Flow straightener

The flow distribution in the pulse tube plays an important role to efficiently transform acoustic power into the cooling power. Since a gas packet acts like a compliant piston in the pulse tube, the flow distribution should be uniform to increase its efficiency. If the flow transition in the front and back sides of the pulse tube makes a non-uniformity in radial direction of flow distribution, the compliant gas piston becomes a distorted shape. This phenomenon leads to an undesirable exergy loss, and the performance of the PTR is considerably degraded. The flow distribution in slit-type heat exchanger is also considered because the flow path of each slit is not mutually connected. The flow non-uniformity in slits would deteriorate the heat exchanger performance, which also leads to the degradation of PTR performance.

A flow straightener is used to overcome these problems where the flow non-uniformity is unavoidable. The flow straightener is made of stacked mesh screens, and the configuration should be optimized. The flow straighteners are located in the front of the aftercooler, both sides of the pulse tube, and between the WHX and the inertance tube. In this paper, the commercial CFD package, Fluent, is utilized to

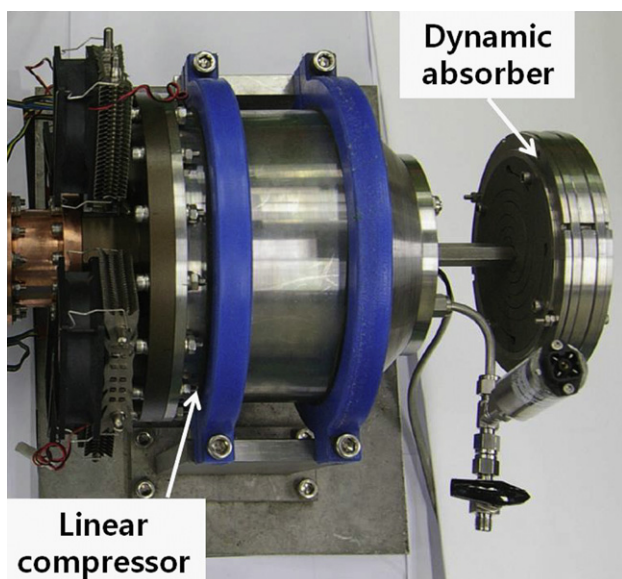


Fig. 2 – Fabricated linear compressor.

Table 1 – Specification of slit-type heat exchangers.

Slit-type heat exchanger	Inlet height (mm)	Outlet height (mm)	Thickness of the slit (mm)	Number of slits	Length (mm)	Lifted heat (W)
Aftercooler (Air-cooled)	26	14	0.35	20	112	400
CHX	14	5	0.30	20	25	–
WHX (Air-cooled)	4.5	4.5	0.30	16	35	25

design the configuration of flow straighteners, and our CFD model (Ki and Jeong, 2010c) is used for optimization of the flow straightener. The structures of slit-type heat exchangers shown in reference (Ki and Jeong, 2010c) are used for the CFD simulation and the grid test has been done. The second-order upwind scheme is used for the convective terms in the momentum equations and the residuals (1×10^{-4}) are defined as the convergence criteria in Fluent. Table 2 (a) presents the results of the grid test in the aftercooler. The time step size of the CFD simulation is determined by using the reference (Lyulina, 2004). The average axial velocity of the working fluid at the flow straightener exit is used to confirm the effect of these various grid size. The minimum grid size producing the identical average velocity is used in this CFD simulation. The length and number of the mesh screens are determined by 3-D simulation as shown in Table 3. The flow uniformity is confirmed by this CFD simulation, and the optimized flow straighteners are inserted in PTR.

2.5. Compressible volume

Compressible volume is an empty volume between the cylinder and the aftercooler. If the hydraulic work generated from the linear compressor is not directly transferred to the aftercooler, compressible volume may be considered as an unnecessary space. However, recent researches show that the compressible volume is useful and indispensable for

matching the impedance between the compressor and the cold part of the PTR. It can actually amplify the pressure amplitude in front of the aftercooler (Sun et al., 2009; Dietrich and Thumm, 2010). The Stirling-type PTR without compressible volume was designed in our previous research. Since the volume of the PTR was smaller than the volume of the back side of the piston, the neutral position of piston was shifted to back during the operation. This shift has resulted in the decrease of the maximum piston amplitude. Also, the efficiency of the PTR was very low because the absence of the compressible volume restricted the control of the impedance between the linear compressor and the cold part. Therefore, a compressible volume is considered in developing an efficient Stirling-type PTR. Care should be taken to accurately estimate the required compressible volume.

3. Modeling and configuration optimization of Stirling-type PTR

In order to design the overall configuration of the Stirling-type PTR and predict the performance, 1-D numerical simulation based on thermal and fluid dynamic equations is utilized. The simulation integrates the behavior of linear compressor and the losses of each component. The programs such as SAGE (Gedeon, 1995) and DeltaE (Ward and Swift, 1996) have been widely used in simulation of PTR. Using those simulation

Table 2 – Test results of grid and time step in simulation.

(a) CFD ^a									
Mesh number					Average axial velocity (m/s) ^b				
731,100					−1.3				
1,100,000					−1.27				
2,005,815					−1.13				
2,715,288					−1.1				
3,606,662					−0.96				
4,308,120					−0.98				
5,084,910					−0.98				
(b) Simulation code of PTR									
Time step	4	5	6	7	9	10	11	12	16
Grid	30	30	30	30	30	30	30	30	30
Cooling capacity (W)	0.51	6.89	7.45	7.33	7.76	7.91	7.95	8.05	8.04
Grid	5	7	10	12	14	15	20	25	30
Time step	12	12	12	12	12	12	12	12	12
Cooling capacity (W)	2.63	6.24	7.6	7.8	7.94	8.02	8	8.07	8.05

a This table shows the results of the grid test at the aftercooler.

b At the used time step, the working fluid flowing from the regenerator to the compressor has the maximum average velocity among the time steps of one cycle.

Table 3 – Specification of flow straightener.

Location	Thickness	Number of mesh
Front of aftercooler	6 mm	# 100 (copper)
Front of pulse tube	3 mm	# 100 (copper)
Back of pulse tube	3 mm	# 100 (copper)
Back of WHX	3 mm	# 100 (copper)

codes, the performance can be predicted with various operating conditions and Stirling-type PTR of different geometry.

3.1. Governing equations

The working fluid of the Stirling-type PTR can be described with the continuity, momentum, and energy equations as follows.

$$\frac{\partial \rho A}{\partial t} + \frac{\partial \rho u A}{\partial x} = 0 \quad (2)$$

$$\frac{\partial \rho u A}{\partial t} + \frac{\partial u \rho u A}{\partial x} + \frac{\partial P}{\partial x} A - F A = 0 \quad (3)$$

$$\frac{\partial \rho e A}{\partial t} + P \frac{\partial A}{\partial t} + \frac{\partial}{\partial x} (u \rho e A + u P A + q) - Q_w = 0 \quad (4)$$

After the physical properties of working fluid in Stirling-type PTR are estimated from these equations, the enthalpy and the expansion work at the cold side of pulse tube are calculated. Then, the losses transferred to the CHX are considered with the enthalpy and the expansion work, and the cooling capacity of Stirling-type PTR is finally predicted. Therefore, the accurate estimation of the physical property and losses is important to accurately predict the cooling capacity. This means that the pressure drop and the loss mechanism generated in the cold part should be properly identified. F is formulated in terms of friction factor f and local loss coefficient K as the following equation (5).

$$F = - \left(\frac{f}{D_h} + \frac{K}{L} \right) \frac{\rho u |u|}{2} \quad (5)$$

In most previous researches, the friction factor and the local loss coefficient were presented as empirical functions. However, the pressure drop of the regenerator under oscillating flow and pressure conditions cannot yet be accurately estimated. This, as a result, leads to somewhat inaccurate performance prediction of the PTR because hydraulic conditions of the working fluid in the regenerator are the most important factor to simulate the performance of the PTR. Therefore, the friction factor used in the simulation code should be adjusted by a correction factor obtained from experimental results. The local loss coefficient expresses the pressure drop at the sharp-edged or bended part, which cannot be ignored. The asymmetric characteristic of pressure drop amplitude according to flow direction can unnecessarily generate a DC flow in the PTR (Gedeon, 1997). For accurately considering the local loss coefficient and suppressing the DC flow, we use the local loss coefficient developed for the slit-type heat exchanger (Ki and Jeong, 2010b). In other parts, appropriate local loss coefficients are selected from the reference (Kays and London, 1984).

Several irreversible losses are generated in regenerator and pulse tube. The parasitic heat due to these losses is transferred to CHX. The losses of pulse tube are composed of the conduction loss of gas and solid wall, the shuttle loss, the second-order flow loss, and the free convection loss as the following Eq. (6).

$$Q_{\text{pulse tube loss}} = Q_{\text{conduction}} + Q_{\text{shuttle}} + Q_{\text{2nd-order}} + Q_{\text{convection}} \quad (6)$$

Among them, the losses in the pulse tube are still difficult to identify because the accurate measurement and separation of each loss according to its origin are difficult. The previous researches presented the methods for predicting the shuttle loss, the second-order flow loss, and the free convection loss using some analytic and simulation models (Jung and Jeong, 2003; Lee et al., 1995; Hollands, 1973). These methods are used in the simulation code to optimally design configuration of the PTR and predict the performance. However, since the predicting methods have some inaccuracy, the presented methods need some calibrations to improve the accuracy. For increasing the accuracy of simulation code, correction factors are used in the friction factor of regenerator and the losses of pulse tube. The correction factors are determined by the results obtained from additional hydraulic tests.

3.2. Design of initial configuration and determination of correction factors

The approximate configuration of Stirling-type PTR should be designed before running the simulation code. Since there are too many possible combinations in the selection of geometric parameters such as diameter and length of regenerator and pulse tube, using all of the possible combinations in simulation code is not recommended. As a starting point, the approximate design method in the reference (Pfotenhauer et al., 2008) is used, and the diameter and the length of regenerator and pulse tube are roughly determined as 2.54 cm and 15 cm, respectively. After then, the optimization process is executed in the simulation code based on the range of the values.

Experiments are executed for determining the correction factors. After fabricating the regenerator based on the simulation result, the pressure drop of regenerator is measured in the steady flow condition. The experimental result is compared with the pressure drop used in the simulation code. The difference between the calculated and the measured pressure drops is adjusted by the correction factor. Fig. 3 shows the calibration result of the friction factor. Since the regenerator used in the calibration process has the same tube diameter and mesh screens with the finally designed regenerator, the correction factor is not changed during the design process.

To determine the correction factors of three losses existing in the pulse tube, three additional experimental results of the previously fabricated Stirling-type PTRs are used (Ki and Jeong, 2010a, 2010c). The Stirling-type PTRs had been optimized using the simulation code which was adjusted by the corrected friction factor. These PTRs had different operating conditions and configurations. After measuring the cooling capacity of the Stirling-type PTRs, the results are compared with the predicted cooling capacity of the previous simulation code. Fig. 4 shows the predicted cooling capacity and the

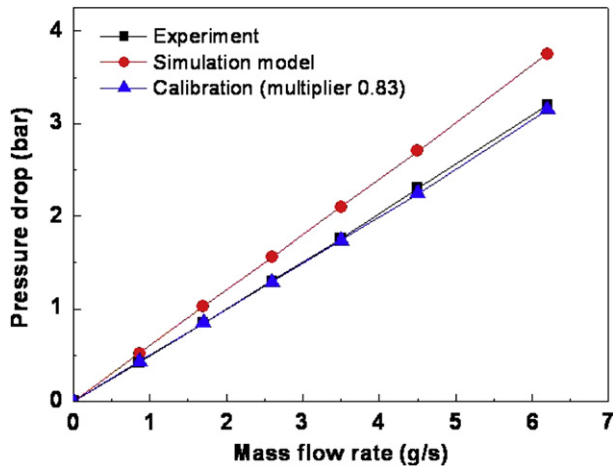


Fig. 3 – Calibration of pressure drop in regenerator.

measured cooling capacity of three different PTRs. To match the predicted cooling capacity with the measured cooling capacity, the correction factors of the losses are adjusted in the simulation code. Finally, the correction factors are found, and they match the predicted cooling capacity and the measured cooling capacity with smallest error. The correction factors of the shuttle loss, the second-order flow loss, and the free convection loss are 1.2, 1.2, and 1.4, respectively. Most simulation codes have useful logics and methods to predict the losses and pressure drop. However, this calibration process should be essentially accompanied to accurately design the configuration and predict the performance of the Stirling-type PTR.

3.3. Result of design process

The Stirling-type PTR in the adjusted SAGE simulation code is composed of linear compressor, compressible volume, flow straighteners, slit-type heat exchangers, regenerator, pulse

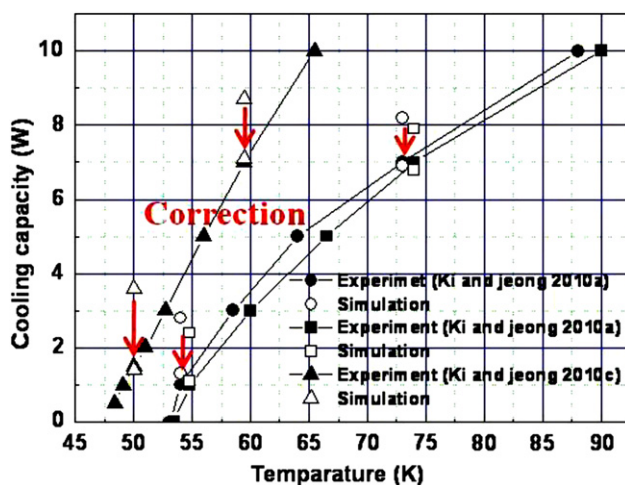


Fig. 4 – Predicted and measured cooling capacity for calibration of the losses in the pulse tube (Arrows show points used in the correction process).

Table 4 – Geometric parameter and operating condition of Stirling-type PTR.

<i>Specifications of the PTR</i>	
Compressible volume	57.15 mm (O.D.), 38.4 mm (L)
Regenerator	31.75 mm (O.D.), 78 mm (L), 0.69 (e_v)
Pulse tube	12.7 mm (O.D.), 154 mm (L)
Inertance tube 1	4.76 mm (O.D.), 0.42 m (L)
Inertance tube 2	6.35 mm (O.D.), 2.47 m (L)
Reservoir volume	1000 cm ³
Heat exchangers	Slit-type
<i>Operating condition and the result</i>	
Frequency	58.7 Hz
Charging pressure	3.4 MPa

tube, inertance tube, and reservoir. After approximate geometric configuration is set up, the optimization process is executed carefully. The optimization process means finding the exact configuration and operating condition of the Stirling-type PTR to efficiently achieve the required cooling capacity. The process is automatically executed in the feasible ranges of the configurations previously determined. Table 2(b) presents the difference of the simulation result according to various grid sizes of each component and time steps. The default value (6.132×10^{-13}) is set as the convergence tolerance of the solver in the simulation code and the estimated cooling capacity is used to confirm the effect of various grid sizes and time steps. From these tests, the grid size and time step selected for the optimization process are 15 and 12, respectively. The finally designed configuration and the operating condition of the Stirling-type PTR are shown in Table 4.

4. Experimental verification

4.1. Experimental setup

Fig. 5 shows the fabricated Stirling-type PTR. The aftercooler and the WHX are made of oxygen-free copper because the parts are brazed. For air-cooling, CPU coolers and aluminium fins are installed on the surface of the aftercooler and the WHX, respectively. An E-type thermocouple measures the exit

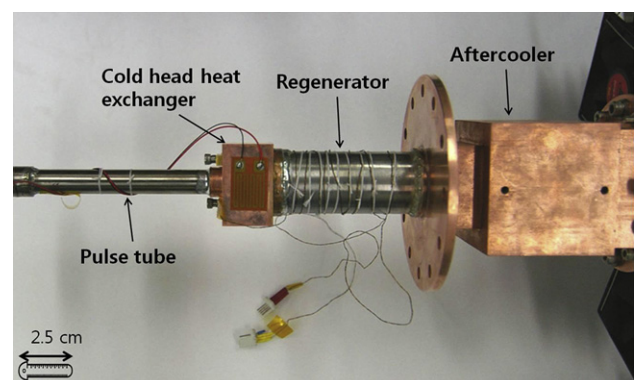


Fig. 5 – Configuration of fabricated Stirling-type PTR.

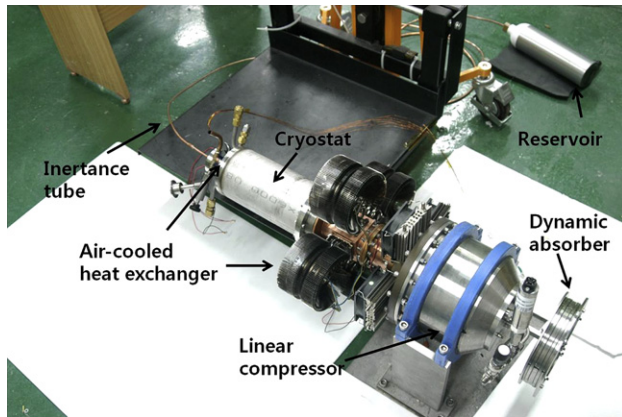


Fig. 6 – Experimental apparatus.

surface temperature of aftercooler for the comparison of the experimental result with the simulation result. The CHX is made of oxygen-free copper. Two silicon diode sensors (DT-670D, Lakeshore) are mechanically mounted using clamp provided by manufacturer, and a heater is attached on the surface of the CHX to measure the cooling capacity.

The regenerator is fabricated by filling 316 stainless steel wire meshes (# 400) into stainless steel tube of 31.75 mm outside diameter (O.D.). In the simulation code, the porosity of the regenerator is calculated by Eq. (7) and is 0.590.

$$e_v = 1 - \frac{\pi D_w \sqrt{\lambda^2 + D_w^2}}{4\lambda^2} \quad (7)$$

However, the actual porosity of the fabricated regenerator is obtained from the weight difference between the filled and empty regenerator as the following Eq. (8), and the measured porosity is 0.694.

$$e_v = 1 - \frac{4M_m}{\rho_m \pi D_r^2 L_r} \quad (8)$$

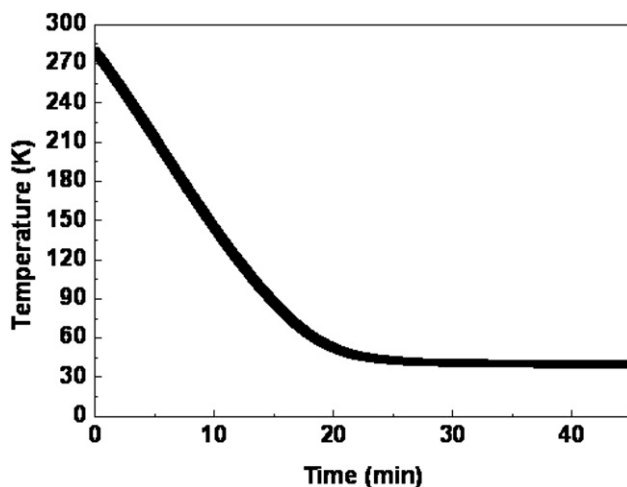


Fig. 7 – Cooling down curve of Stirling-type PTR.

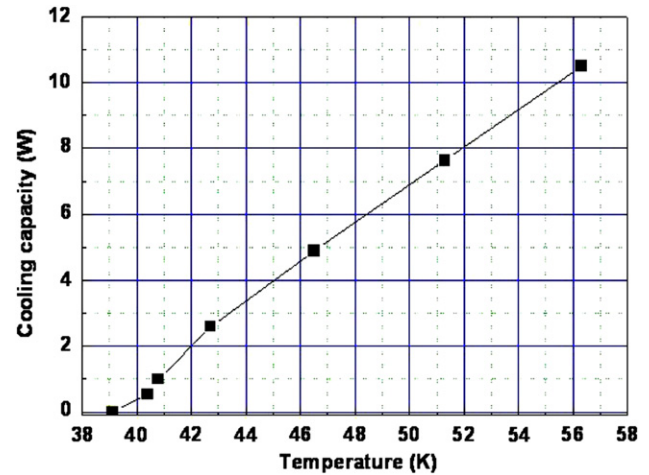


Fig. 8 – Cooling capacity of Stirling-type PTR.

The fabricated regenerator has a different void volume with the regenerator calculated by the simulation code, and this results in the change of pressure amplitude and phase of the working fluid in the CHX. Therefore, actual the hydraulic condition of the working fluid optimized from the simulation code can be different from the previously designed one. The configuration of the Stirling-type PTR is adjusted again from

Table 5 – Comparison between model prediction and experimental measurement.

	Prediction	Measurement
Cold-end heat exchanger temperature (K)	51.3	51.3
Cooling capacity (W)	8.8	7.7
Compressor input power	410	419
Aftercooler temperature (K)	309	312
Regenerator type	# 400 stainless steel mesh	
Carnot efficiency (%)	10.8	9.4

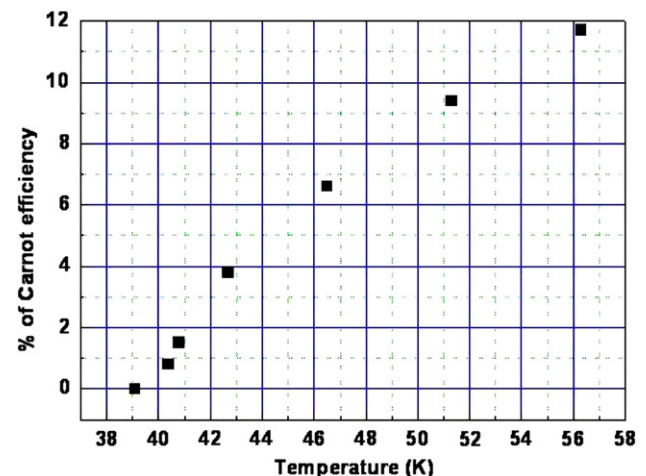


Fig. 9 – The Carnot efficiency of Stirling-type PTR.

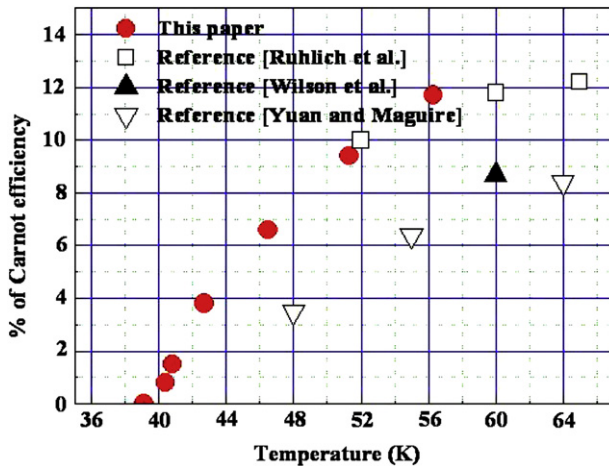


Fig. 10 – Comparison of the Carnot efficiency with other PTRs.

the simulation code using the actual porosity measured from the fabricated regenerator.

Flow straighteners are made by copper wire mesh of #100, and inserted to the predetermined spaces in the compressible volume, CHX, and WHX. Each flow straightener is then finally attached at the rim by soldering. The compressible volume is fabricated by usual copper, and the inertance tube is designed and fabricated by a double-diameter type (Marquardt and Radebaugh, 2000) to delicately control the phase of the working fluid in the CHX. All measured data are collected by NI-board (SCXI-1000, 1125, and 1328) and handled by the LABVIEW 8.2 program. All sensor lines and heater wires are thermally anchored on the regenerator and the pulse tube to reduce further thermal conduction loss through the instrumentation wires. Additionally, 10 multi layer insulation (MLI) sheets supported by G10 tubes are installed to reduce radiation heat invasion. The MLI sheets are tightened around the G10 tubes with threads (Dental Floss, Ranir).

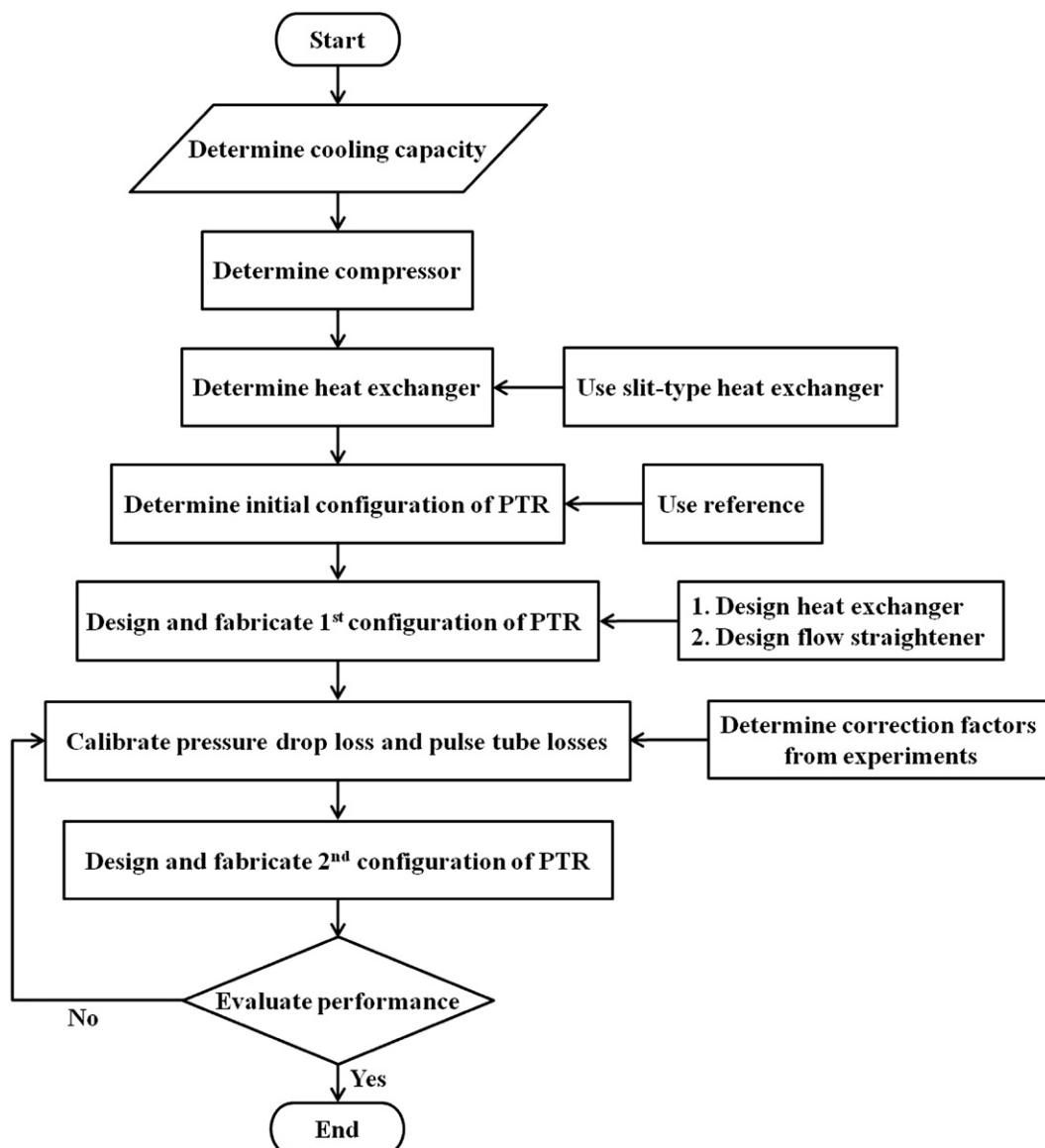


Fig. 11 – Flow chart of design process.

4.2. Experimental results

Fig. 6 shows the experimental apparatus which is located horizontally. The Stirling-type PTR charged by helium gas at 3.38 MPa operates at 58.7 Hz. Fig. 7 shows the cool down curve of the Stirling-type PTR. The no-load temperature is 39.1 K with maximum electrical power of 419 W and it takes about 30 min to cool down up to this temperature. The measured exit surface temperature of the aftercooler is 312 K, which is 3 K higher than the designed exit surface temperature. The result indicates that the slit-type heat exchanger is designed and properly fabricated with relatively small error.

As shown in Fig. 8, the cooling capacity is measured for judging the accuracy of the design process of the Stirling-type PTR. The cooling capacity of 7 W at 50 K is achieved, and it shows the small difference compared to the target of 8 W at 50 K. As an example, Table 5 shows the comparison between the model prediction and the experimental measurement. The design process described in this paper is considered to have sufficient accuracy. From the measured cooling capacity, the efficiency of the Stirling-type PTR is evaluated with respect to the Carnot efficiency. In this paper, the electrical power of the linear compressor is used for calculating the Carnot efficiency because the efficiency of the linear compressor must be also included in calculating the total efficiency of Stirling-type PTR. Fig. 9 shows the curve of the Carnot efficiency calculated from Eq. (9).

$$\% \text{ of Carnot efficiency} = \frac{\frac{Q_{\text{ref}}}{W_{\text{electrical input}}}}{\frac{T_{\text{C}}}{T_{\text{amb}} - T_{\text{C}}}} \quad (9)$$

The fabricated Stirling-type PTR has the Carnot efficiency of 9.4% and 11.7% at 51.3 K and 56.3 K, respectively and the result is compared with other PTRs (Ruhlich et al., 2001; Wilson et al., 2007; Yuan and Maguire, 2005). Fig. 10 shows the comparison, and we can evaluate that the fabricated Stirling-type PTR has a comparable efficiency with other efficient PTRs. The Stirling-type PTR in this paper is optimized from the design process to efficiently convert electrical power to mechanical power for producing the required cooling effect. Fig. 11 shows the flow chart that summarizes the design process.

5. Discussion

From the step-by-step design process explained in this paper, an efficient Stirling-type PTR is designed and fabricated. As a result, we can confirm the accuracy and usefulness of the design process. However, additional experiments are required for determining the correction factors to predict accurately pressure drop and losses. To avoid the additional experiments, we need to propose more systematic approach to analyze the pressure drop characteristics of regenerator and estimate the losses in pulse tube. These analytic efforts may be made by measuring the actual physical parameters of working fluid in various configurations of PTR. It is expected that the energy flow measurements at several prime locations

of PTR under realistic operational conditions shall clarify and better understand this issue.

6. Conclusion

The design process divided into three sub-topics has been presented in this paper and each process is carefully explained by the step-by-step procedure. The procedures are as follows: 1) specific design and fabrication of each component, 2) modeling and configuration optimization of Stirling-type PTR, and 3) experimental verification. From this design process, a Stirling-type PTR is designed and fabricated for effectively obtaining the cooling capacity of 8 W at 50 K. The fabricated Stirling-type PTR has the cooling capacity of 7 W at 50 K and the Carnot efficiency of 11.7% at 56.3 K. This design process has a good accuracy and it is considered to be a proper design methodology for an efficient Stirling-type PTR. The proposed design procedure is useful to help researchers easily develop efficient Stirling-type PTRs for wide acceptance in cryogenic cooling applications.

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REFERENCES

- Dietrich, M., Thummes, G., 2010. Development of a high-power two-stage high frequency pulse tube cooler for refrigeration at 25 K. In: To Be Presented at the International Cryogenic Engineering Conference 23 and International Cryogenic Materials Conference 2010.
- Gedeon, D., 1995. SAGE: object-oriented software for cryocooler design. In: Proceedings of 8th International Cryocooler Conference, pp. 281–292.
- Gedeon, D., 1997. DC gas flows in stirling and pulse tube cryocoolers. *Cryocoolers* 9, 385–392.
- Hollands, K.G.T., 1973. Natural convection in horizontal thin-walled honeycomb panels. *Trans. ASME J. Heat Transfer* 95, 439–444.
- Jung, J., Jeong, S., 2003. Surface heat pumping loss in pulse tube refrigerator. *Cryocoolers* 12, 371–378.
- Kays, W.M., London, A.L., 1984. *Compact Heat Exchangers*. McGraw-Hill, New York.
- Ki, T., Jeong, S., 2010a. Optimal design of the pulse tube refrigerator with slit-type heat exchangers. *Cryogenics* 50, 608–614.
- Ki, T., Jeong, S., 2010b. Pressure drop characteristics of slit-type heat exchanger. *Cryocoolers* 16, 201–209.
- Ki, T., Jeong, S., 2010c. Stirling-type pulse tube refrigerator with slit-type heat exchangers for HTS superconducting motor. *Cryogenics* (Available online).
- Kittel, P., 2008. Comparison of two empirical cryocooler models. In: Proceedings of the International Cryogenic Engineering Conference and International Cryogenic Materials Conference 2008, pp. 777–780.

- Lee, J.M., Kittel, P., Timmerhaus, K.D., Radebaugh, R., 1995. Steady secondary momentum and enthalpy streaming in the pulse tube refrigerator. *Cryocoolers* 8, 359–369.
- Lewis, M.A., Taylor, R.P., Bradley, P.E., Garaway, I., Radebaugh, R., 2009. Pulse tube cryocooler for rapid cooldown of a superconducting magnet. *Cryocoolers* 15, 167–176.
- Lyulina, I., 2004. Numerical Simulation of Pulse Tube Refrigerators. Eindhoven University Press, The Netherlands.
- Marquardt, E.D., Radebaugh, R., 2000. Pulse tube oxygen liquefier. *Adv. Cryog. Eng.* 45, 457–464.
- Pfotenhauer, J.M., Gan, Z.H., Radebaugh, R., 2008. Approximate design method for single stage pulse tube refrigerators. *Adv. Cryog. Eng.* 53, 1437–1444.
- Richardson, R.N., Evans, B.E., 1997. A review of pulse tube refrigeration. *Int. J. Refrigeration* 20, 367–373.
- Ruhlich, I., Korf, H., Wiedmann, T., 2001. The AIM space cryocooler program. *Cryocoolers* 11, 139–144.
- Sun, D.M., Dietrich, M., Thumm, G., 2009. High-power Stirling-type pulse tube cooler working below 30 K. *Cryogenics* 49, 457–462.
- Taylor, R.P., Nellis, G.F., Klein, S.A., 2008. Optimal pulse-tube design using computational fluid dynamics. *Adv. Cryog. Eng.* 53, 1445–1453.
- Ward, B., Swift, G., 1996. Design Environment for Low-amplitude Thermoacoustic Engines: DeltaE. Los Alamos National Laboratory. LA-CC-93-8.
- Wilson, K.B., Fralick, C.C., Gedeon, D., Yoshida, M., Kawahara, S., 2007. Sunpower's CPT60 pulse tube cryocooler. *Cryocoolers* 14, 123–132.
- Yuan, J., Maguire, J., 2005. Development of a single stage pulse tube refrigerator with linear compressor. *Cryocoolers* 13, 157–163.